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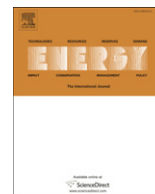
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# Characteristics of LPG-diesel dual fuelled engine operated with rapeseed methyl ester and gas-to-liquid diesel fuels

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## ABSTRACT

A Liquefied Petroleum Gas (LPG)-diesel dual fuelled combustion experimental study was carried out to understand the impact of the properties of the direct injection diesel fuels, such as rapeseed methyl ester (RME) and gas-to-liquid (GTL), on combustion characteristics, engine performance and emissions. The experimental results showed that up to 60% of liquid fuel replacement by LPG was reached while keeping engine combustion variability within the acceptable range and obtaining clear benefits in the soot-NO<sub>x</sub> trade-off. However, the amount of LPG was limited by adverse effects in engine thermal efficiency, HC and CO emissions. LPG–RME showed a good alternative to LPG–diesel dual fuelling, as better engine combustion variability, HC, CO and soot behaviour was obtained when compared to the other liquid fuels, mainly due to its fuel oxygen content. On the other hand, NO<sub>x</sub> emissions were the highest, but these can be balanced by the application of EGR. LPG–GTL dual fuelling resulted in the highest NO<sub>x</sub> emissions benefit over a wide range of engine operating conditions. The high cetane number and the absence of aromatic of GTL are the main parameters for the more favourable soot-NO<sub>x</sub> trade-off compared to LPG–ULSD (ultra low sulphur diesel) dual fuelling.

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## 1. Introduction

Liquefied petroleum gas (LPG) is considered as a promising alternative fuel and has been widely used in transportation due to its environmental and economic benefits [1–3]. Apart from being a lower pollutant, LPG is also desirable as it receives tax concession in Europe [1,4].

The high octane number of LPG makes it suitable for spark ignition engines. In contrast, the low cetane number (CN) of LPG makes it difficult to be used in large proportions in compression ignition engines, mainly due to high cyclic variation [5]. However, the addition of cetane enhancer can improve the performance of LPG fuelled diesel engine operation [5–7].

It has been reported that with the use of in-cylinder injected LPG, improvements in smoke and nitrogen oxides (NO<sub>x</sub>) emissions are evident and in some cases carbon monoxide (CO) emission can be minimised, specifically in high engine load [8]. The presence of the LPG spray combined with diesel fuel into the cylinder promotes diesel atomisation, increasing the velocity of diffusion combustion, which in turn diminishes soot formation [9].

In dual fuel engine operation, the combustion characteristics are affected by both the pilot diesel fuel as ignition source and the primary premixed fuel. For instance, the differences in chemical and physical properties of the pilot fuel directly affect the emissions and performance of the engine mode [7,10,11]. Therefore, the high CN of GTL is expected to improve the engine performance and emissions of LPG-liquid fuel engine operation [12]. The use of high CN fuels could improve the auto-ignition characteristics and extend stable engine operation under LPG-diesel dual fuelling [5].

Similarly, the combustion of biodiesel fuel like rapeseed methyl ester (RME) shows emissions improvements mainly in terms of HC, CO, and soot. The oxygen content of RME (about 10.8%wt) improves fuel oxidation, inhibiting the formation of carbonaceous pollutant species and enhancing their oxidation [13–15]. However, changes in the injection and combustion patterns due to high biodiesel's bulk modulus and oxygen content have been reported which tend to raise NO<sub>x</sub> emissions [10,11,13]. However, it is expected that this drawback can be minimised by the LPG combustion characteristics.

The objective of this work is to investigate the influence of three in-cylinder injected diesel fuels, ULSD, RME and GTL on the combustion characteristics and emission of the LPG-diesel dual fuelled engine. The influence of the exhaust gas recirculation (EGR) was also investigated.

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**Table 1**  
Experimental engine specifications.

Engine specification	Data
Engine cycle	4-stroke DI
Number of cylinders	1
Bore/stroke	98.4 mm/101.6 mm
Connecting rod length	165 mm
Displacement volume	773 cm <sup>3</sup>
Compression ratio	15.5:1
Rated power (kW)	8.6 @ 2500 rpm
Peak torque (Nm)	39.2 @ 1800 rpm
Injection system	Three hole direct injection
Engine piston	Bowl-in-piston

**Table 2**  
Liquid fuel properties.

Property	Method	ULSD	RME	GTL
Cetane number	ASTM D613	53.9	54.7	80
Density at 15 °C (kg/m <sup>3</sup> )	ASTM D4052	827.1	883.7	784.6
Viscosity at 40 °C (cSt)	ASTM D455	2.467	4.478	3.497
50% distillation (°C)	ASTM D86	264	335	295.2
90% distillation (°C)	ASTM D86	329	342	342.1
LCV (MJ/kg)		42.7	37.4	43.9
Sulphur (mg/kg)	ASTM D2622	46	5	<10
Aromatics (%wt)		24.4	~0	0.3
O (%wt)		~0	10.8	~0
C (%wt)		86.5	77.2	85
H (%wt)		13.5	12.0	15
H/C ratio (molar)		1.88	1.85	2.10

## 2. Experimental setup

### 2.1. Engine

Tests were carried out using a naturally aspirated, single cylinder and mechanical direct injection compression ignition diesel engine. An electric dynamometer with a motor and load cell was coupled to the engine and used to motor and load the engine. A detailed engine specification is shown in Table 1 and the schematic diagram of the experimental setup is shown in Fig. 1.

### 2.2. Fuels

Three liquid fuels were used: ULSD, RME and GTL which were provided by Shell Global Solutions UK. The fuel properties are given in Table 2. LPG used in this experiment was provided in a gas cylinder. Propane and butane is generally the main component of LPG, but its actual composition varies widely among countries and depends on season and cost. In the UK, the quality specifications for LPG conformed to BS 4250 which is specified for commercial propane and butane. In this work the LPG used contains 100% propane, and its properties are given in Table 3.

### 2.3. Combustion and emissions facilities

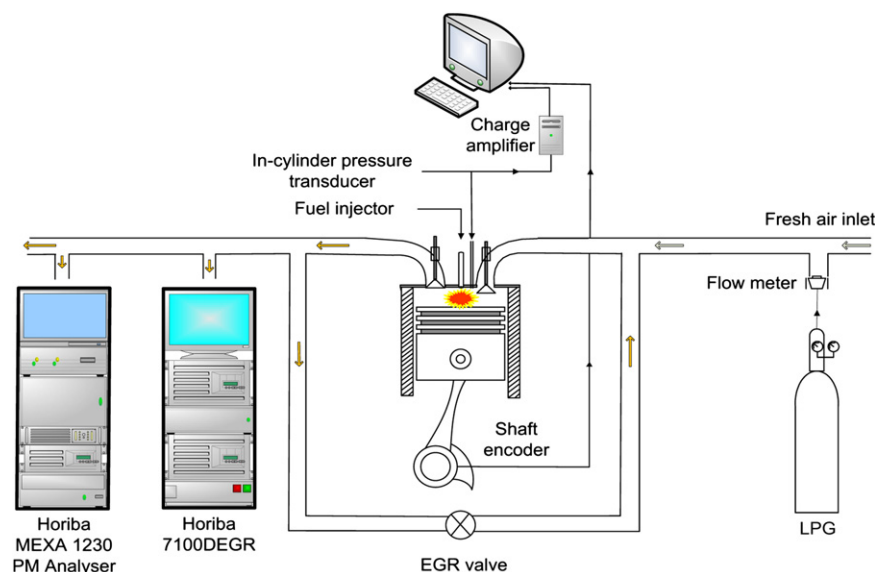
Emissions measurements such as carbon dioxide, carbon monoxide, nitrogen monoxide, nitrogen oxides and gaseous

hydrocarbons were carried out using an HORIBA 7100DEGR emission analyser. Particulate matter (PM) was evaluated using an HORIBA MEXA 1230 PM analyser. In this equipment soot was measured with a diffusion-charging (DC) detector and two flame ionisation detectors (FID) were used for soluble organic material (SOM) measurement. The emissions measurements were carried out three times and an average reading was taken.

The in-cylinder pressure was recorded by using a Kistler 6125B pressure transducer (1% measurement accuracy) mounted flush at the cylinder head and connected via a Kistler 5011 charge amplifier to a National Instruments data acquisition board. A digital shaft encoder was used to measure the crankshaft position. 200 consecutive engine cycles were performed in order to analyse the in-cylinder pressure and rate of heat release. Data acquisition and combustion analysis were carried out using the in-house developed LabVIEW-based software. Output from the analysis of consecutive engine cycles included peak in-cylinder pressure, indicated mean effective pressure (IMEP), coefficient of variation (% COV) of IMEP and peak in-cylinder pressure, rate of heat release (ROHR) and other standard combustion parameters. The COVs are used as criteria for combustion stability.

### 2.4. Engine operating conditions

A modification was made on the intake manifold to allow LPG and fresh air mixing, while the liquid fuel was injected directly in

**Fig. 1.** Schematic diagram of experimental setup.

**Table 3**  
Propane properties.

Property	Propane
Relative density (15.6 °C, 1 atm)	1.5
Boiling point (°C)	−42.1
Latent heat of vaporisation at 15.6 °C (kJ/kg)	358.2
Flammability range (%vol. in air)	2.2–9.5
Auto-ignition temperature (°C)	470
Sulphur (%wt)	0–0.02
LCV (MJ/kg)	46.3
Theoretical air requirement (m <sup>3</sup> /m <sup>3</sup> )	24

**Table 4**  
Engine operating conditions.

Engine load	EGR	LPG concentration
3 and 5 bar IMEP	0 and 20%	0, 0.2, 0.5 and 1%

the combustion chamber. Two different engine loads, 3 and 5 bar IMEP and a constant engine speed (1500 rpm) were used (Table 4). To examine the effect of EGR addition on the LPG–diesel dual fuelled engine operation a 20% EGR concentration was employed. This is expressed as a volumetric percentage of the total air inlet charge, with 0% EGR as the baseline. LPG fed into the combustion chamber was a gas phase with three different levels of flow rates, 0.2, 0.5 and 1% of the total volumetric intake charge air flow rate at the engine inlet. Engine tests were carried out using standard injection timing.

### 3. Results and discussion

#### 3.1. Effect of LPG and EGR for ULSD

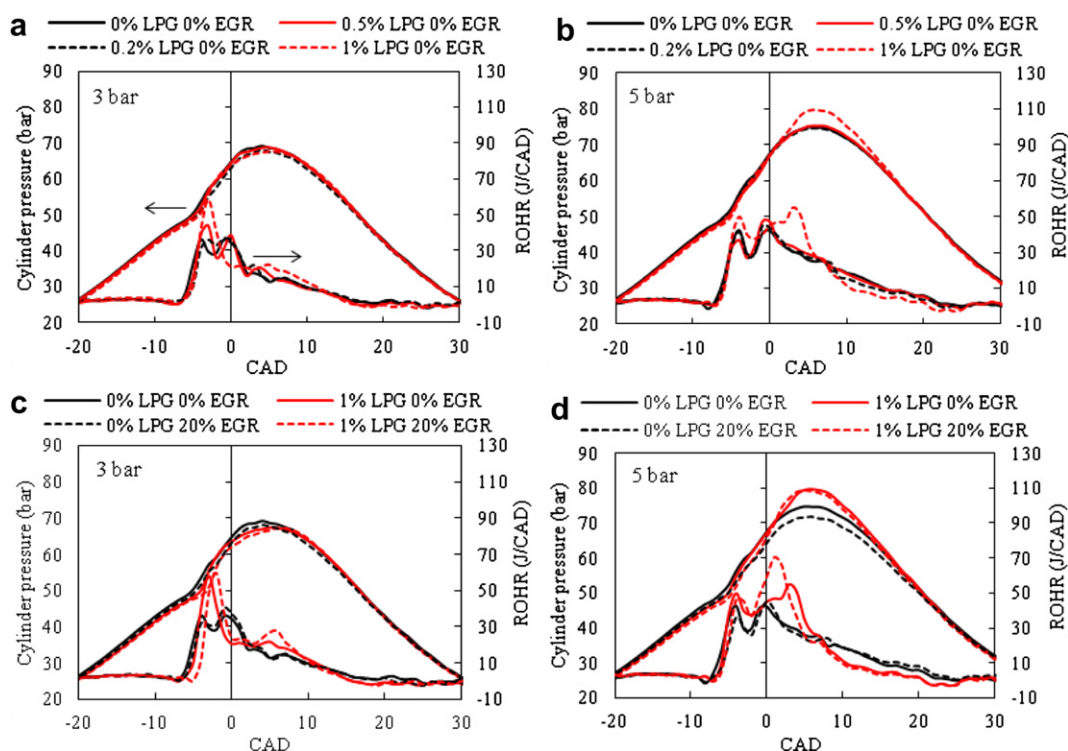
##### 3.1.1. Combustion

Fig. 2 shows the in-cylinder pressure and rate of heat release traces of LPG–ULSD combustion for different percentages of LPG. As the LPG percentage was increased the start of combustion (SOC)

was slightly delayed. The reasons for justifying this delay include the low CN of LPG, the higher heat capacity of the mixture and the lack of oxygen, leading to a longer ignition delay [16]. Moreover, the more LPG concentration in the cylinder the more pronounced was the premixed combustion phase. As LPG is inducted along with the air in the induction stroke, there is sufficient time to mix LPG with air in the cylinder. This leads to the formation of a ready-combustible charge mixture, which is burned in the premixed combustion phase. At low engine load, the premixed combustion phase was dominant in dual fuel engine operation (Fig. 2a). In addition, the unburned LPG at this condition resulted in lower in-cylinder temperature. However, at high engine load the ignition delay was shortened and the liquid fuel injection was extended compared to the low engine load. The higher in-cylinder temperature at high engine load promotes LPG combustion. In addition, the high calorific value of LPG contributes to a higher pressure rise rate and hence in-cylinder pressure with respect to ULSD. Additionally, the diffusion combustion phasing was extended (Fig. 2b).

With EGR addition the SOC was further retarded and the in-cylinder pressure slightly decreased (Fig. 2c and d). The N<sub>2</sub>, CO<sub>2</sub> and H<sub>2</sub>O content within EGR potentially lowers the in-cylinder temperature acting as a heat-absorbing agent due to their high specific heat capacity (thermal effect) [17,18]. The limited oxygen availability in the combustion process due to EGR application ends in poor pre-ignition preparation and longer ignition delay [19,20]. In addition, the combination of EGR and LPG resulted in even longer ignition delay because of the low CN of LPG.

Up to 60% diesel fuel substitution on a mass basis was obtained when 1% LPG was inducted at low engine load condition (Fig. 3a). A higher liquid fuel replacement level was achieved at low engine load because the percentage of LPG addition was kept constant while the amount of liquid fuel was low. However, the substitution of liquid fuel by LPG and EGR should be limited due to misfire and high combustion variability, as a consequence of the low CN of LPG and the insufficient oxidant.



**Fig. 2.** Cylinder pressure and rate of heat release for ULSD. a) 3 bar IMEP, b) 5 bar IMEP, c) effect of LPG and EGR at 3 bar IMEP, d) effect of LPG and EGR at 5 bar IMEP.

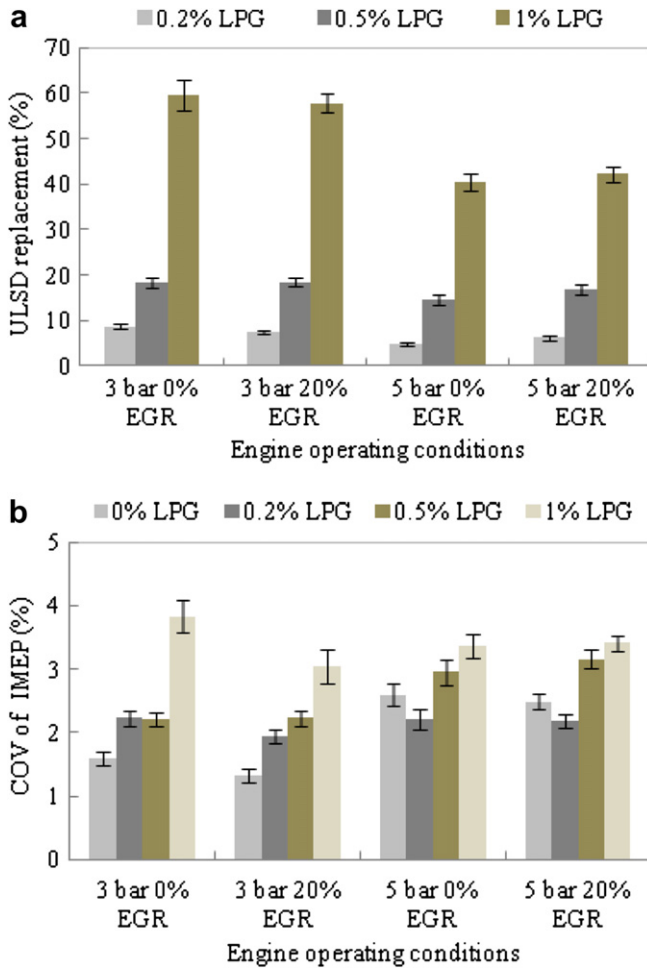


Fig. 3. Liquid fuel replacement and COV of IMEP for ULSD.

Combustion stability, which is indicated by the coefficient of variation (COV) of indicated mean effective pressure (IMEP), was in the acceptable range (lower than 5%) in all the tested conditions (Fig. 3b) [21]. An increasing trend is observed with the increase in LPG quantity, limiting the use of LPG-diesel dual fuel operation. A non-uniform mixture may lead to unstable combustion under a high LPG fraction. A higher quantity of LPG reduces the amount of combustion air and the in-cylinder injected diesel fuel, leading to misfire which can produce relatively high cycle-by-cycle variation [9,22].

The reasons to explain the decreased brake thermal efficiency (Fig. 4) in LPG-diesel dual fuelled engine operation mode can be the reduction in volumetric efficiency, incomplete combustion, reduced flame propagation and an abundant presence of residual gases [4,9,23]. Furthermore, the reduction in the amount of liquid fuel used to initiate the combustion has adverse effects on the quality of liquid fuel spray. This produces poor liquid fuel preparation and atomisation due to a slow development of fine droplets which affects the mixture combustion process [21]. These factors led to a shift of combustion to the expansion stroke, hence producing less useful work [16]. With EGR addition, it enriches the inducted LPG-air charge mixture increasing slightly the intake temperature which could enhance LPG combustion [4,23]. Therefore, the engine thermal efficiency was not further deteriorated compared to 0% EGR.

### 3.1.2. Emissions

An increase in HC and CO emissions was observed in LPG-diesel dual fuelling especially in 1% LPG at low engine load condition

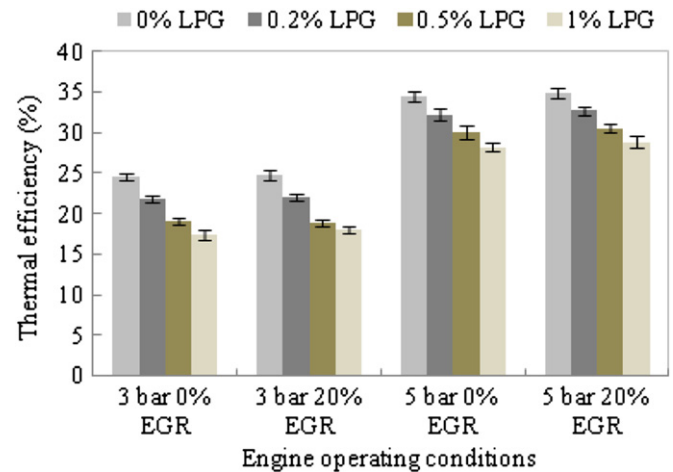


Fig. 4. Brake thermal efficiency for ULSD.

(Fig. 5). Reduced volumetric efficiency, poor mixing quality, flame quenching, and rich premixed mixture at this condition led to incomplete combustion [4,21]. The lower overall lambda, for instance at low engine load, from 3.5 (0% LPG) to 2.6 (1% LPG) also deteriorates the HC and CO emissions levels. In addition, the lower in-cylinder temperature compared with 0% LPG addition and the high LPG auto-ignition temperature (Table 3) also led to poor mixture oxidation. Unburned mixture remains in the cylinder due

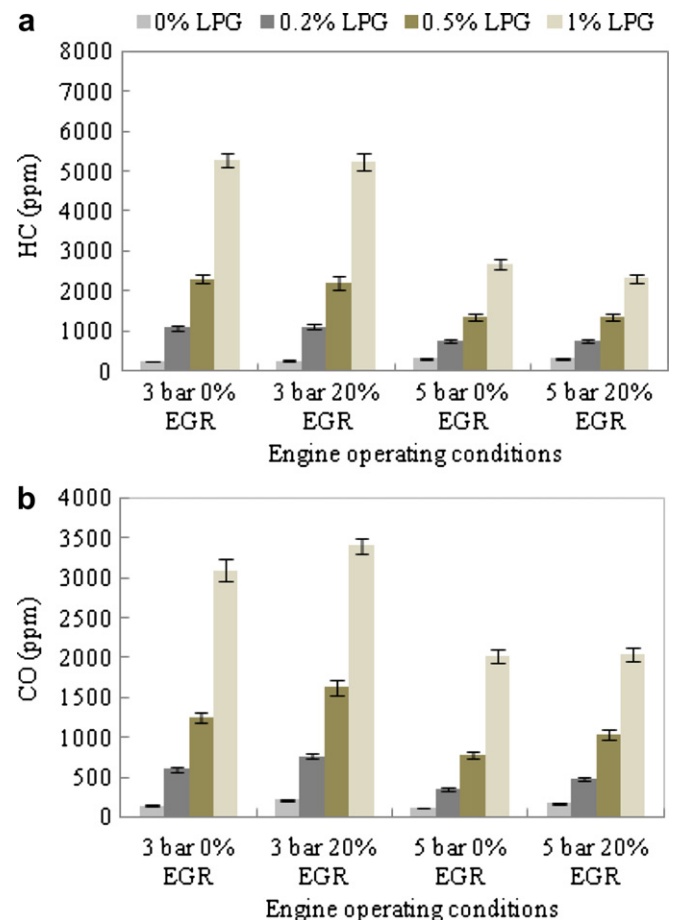


Fig. 5. Hydrocarbon and carbon monoxide emissions for ULSD.



to the slow combustion velocity, which cannot propagate well to combust all the mixture [23]. At high engine load the trend was similar to low engine load, but the HC and CO penalty was lower because of the higher combustion temperature, which promotes a better rate of mixture decomposition and oxidation. Therefore, a relatively constant  $\text{NO}_x$  emission in LPG-diesel dual fuelling was observed compared to diesel only combustion (Fig. 6).

At low engine load although there was a relatively high energy released in the premixed combustion phase, the LPG and EGR additions retard the SOC, inhibiting  $\text{NO}_x$  formation [17,19,23,24]. The reduction of the oxygen due to EGR dilution led to extension of flame region as a result of the mixing between oxygen and fuel was decelerated. This bigger size of flame region absorbed heat from their surrounding thus lowering the flame temperature [24]. Lower in-cylinder pressure with EGR application (Fig. 2c) suggests that the combustion temperature decreases as a result of the dilution and thermal effect of EGR leading to reduced  $\text{NO}_x$ .

At high engine load the higher in-cylinder temperature favours  $\text{NO}_x$  emissions formation [17]. In addition, when 1% LPG was inducted,  $\text{NO}_x$  emissions were higher than those obtained with ULSD only combustion. This suggests that fuel injection modification and a limitation on LPG addition are required to avoid this  $\text{NO}_x$  penalty at high engine load.

Soot is mainly produced in the locally liquid fuel rich regions which are formed in the combustion process [15,25]. Therefore, when LPG was inducted into the cylinder it replaced some quantity of the liquid fuel decreasing the number of locally liquid fuel rich regions, producing less soot (Fig. 6). Combining EGR and LPG,  $\text{NO}_x$  emissions can further be improved while keeping the soot levels low when compared with the diesel only combustion, specifically at low engine load with 1% LPG addition. It is suggested that at this condition the local in-cylinder temperature is considerably reduced, hence inhibiting soot formation. Lean-liquid fuel regions and low air entrainment because of the presence of gaseous fuel also hampered the formation of soot precursor [25].

Soluble organic material (SOM) in PM mainly comes from unburned or partially oxidised hydrocarbons from the parent fuel and lubricating oil, which are adsorbed/condensed onto soot particles [26]. The proportion of SOM in the total PM is defined as soluble organic fraction (SOF) and can be expressed as follows:

$$\text{SOF} = \text{SOM} / (\text{SOM} + \text{SOOT})$$

From Fig. 7a, it was obtained that as LPG addition was increased, SOM reduced, even though unburned total hydrocarbons emissions

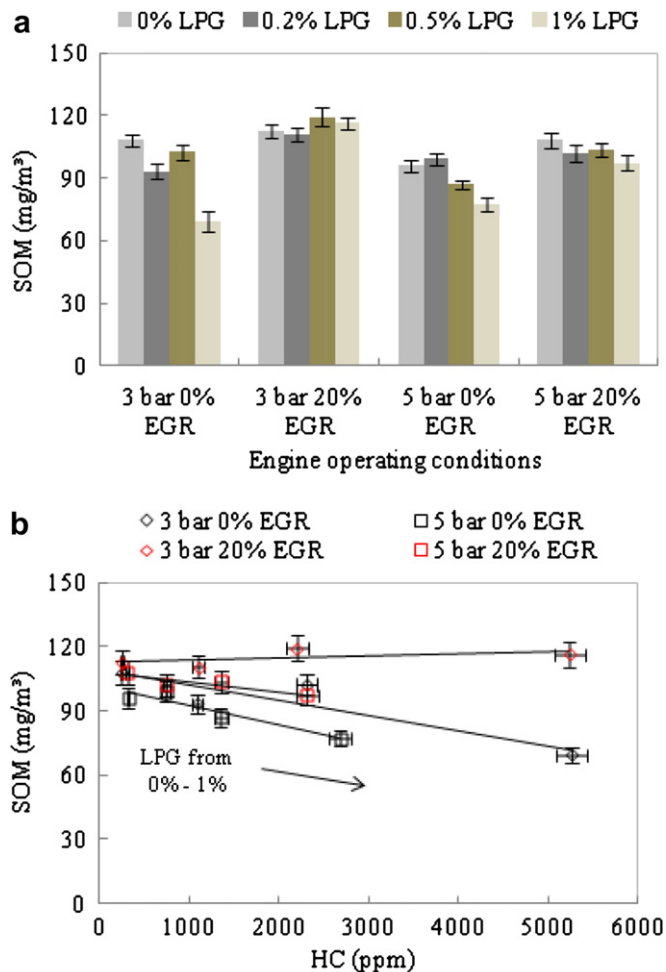


Fig. 7. Soluble organic material (SOM) and HC-SOM correlation for ULSD.

were higher (Fig. 7b). The significant reduction of soot with LPG limits the soot surface area in which organic material can be adsorbed/condensed, hence reducing SOM. However, the high concentration of hydrocarbons with LPG addition promotes their adsorption and condensation onto this limited soot, slightly increasing SOF (Fig. 8). This SOF increment is limited as most of the

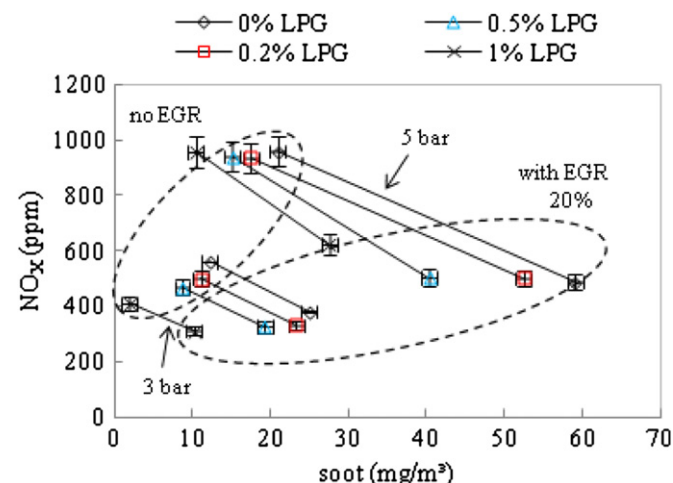


Fig. 6. Soot- $\text{NO}_x$  trade-off for ULSD.

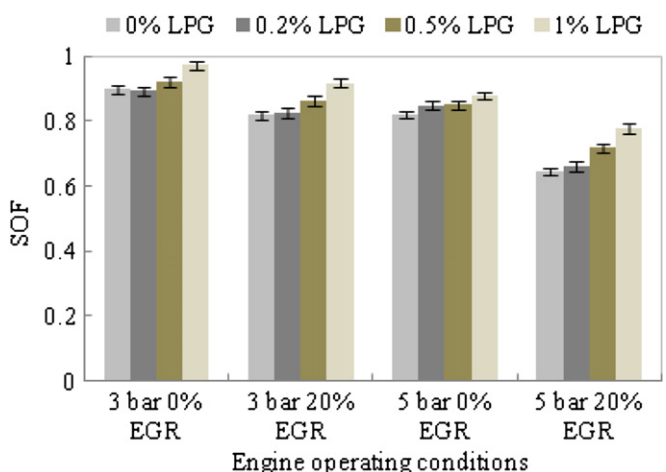


Fig. 8. Soluble organic fraction (SOF) for ULSD.

engine-out gaseous hydrocarbons in LPG combustion might be unburned LPG and other light hydrocarbons. These hydrocarbons have a low dew point restricting their adsorption/condensation onto soot particles.

The engine operating conditions also affect the quantity of SOM. In the case of high engine load where lambda was lower, the in-cylinder temperature was high and hydrocarbons emissions were lower compared to low engine load hence the SOM was reduced [27]. In the case of EGR application, the SOM was higher compared to the case without EGR, while the SOF was lower. When EGR was added, soot concentration significantly deteriorated, while hydrocarbons emissions were similar with respect to no EGR application (Fig. 5a). As a consequence, there is more available soot surface in which hydrocarbons could be adsorbed or condensed, increasing the total SOM. However, the organic material per particle was reduced (SOF), as there were more soot particles while the hydrocarbons concentration remained similar.

### 3.2. Effect of liquid fuels

#### 3.2.1. Combustion

Fig. 9 depicts the in-cylinder pressure and heat release rate of the different liquid fuels at 0 and 1% LPG. Similar results among the different liquid fuels were obtained when 0.2 and 0.5% of LPG were added but these combustion patterns are not shown.

The premixed combustion phase with GTL was low because of the reduction in ignition delay derived from its high CN (Table 2). Therefore, most of the GTL was burned in the diffusion combustion phase, extending the combustion duration. However, the SOC of GTL was comparable to that obtained with ULSD. This similar SOC, despite the shorter ignition delay of GTL, is due to the delay in the start of injection (SOI). This delay comes from the low density and high compressibility of GTL (Table 2). These GTL combustion

characteristics resulted in lower in-cylinder pressure. On the other hand, higher density and bulk modulus of RME make it less compressible so that the pressure in the injector can develop faster to reach the required injection pressure [28,29]. This led to an earlier start of fuel injection and combustion, raising the in-cylinder pressure.

The addition of LPG and EGR changed the combustion patterns with respect to the original ones (e.g. LPG–diesel). When 1% LPG was inducted in GTL dual fuel combustion, the low quantity of GTL (i.e. reduction in the auto-ignition properties of the mixture) cannot compensate the injection delay, producing a retard at the start of ignition. With EGR the SOC for the three fuels was retarded. The higher heat capacity of the mixture and lower oxygen availability are the barriers for early ignition. The delay in the SOC was more noticeable in the case of GTL, as it was explained previously in 1% LPG.

Liquid fuel replacement and combustion variability from cycle-by-cycle for the different fuels are shown in Fig. 10. RME and ULSD showed comparable engine thermal efficiencies (Fig. 11). These similar thermal efficiencies and the lower calorific value of RME led to a higher quantity of injected RME compared with ULSD resulting in lower liquid fuel replacement (Fig. 10a).

In the case of GTL dual fuelling, GTL has higher calorific value compared with ULSD. Therefore, it is expected that a smaller quantity of GTL is required to achieve the same output engine power than with ULSD. However, at low engine load the engine thermal efficiency (explained later) for GTL dual fuelling was lower compared with ULSD dual fuelling hence more liquid fuel was used resulting in lower liquid fuel replacement.

The addition of LPG caused an increase in the cyclic variability for both low and high engine load (Fig. 10b). However, the LPG penalty to the cyclic variability was more pronounced in the case of low engine load due to less effective combustion of more diluted in-

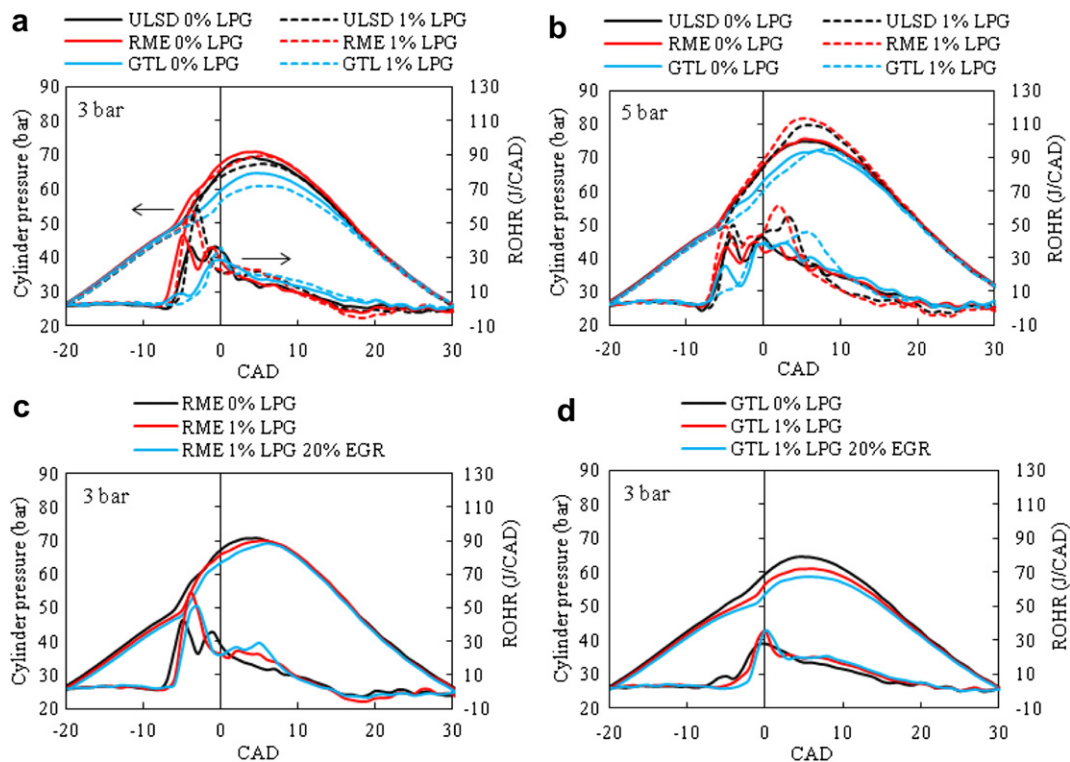


Fig. 9. Cylinder pressure and rate of heat release. a) effect of different liquid fuels at 3 bar IMEP, b) effect of different liquid fuels at 5 bar IMEP, c) effect of LPG and EGR on LPG–RME dual fuel combustion, d) effect of LPG and EGR on LPG–GTL dual fuel combustion.

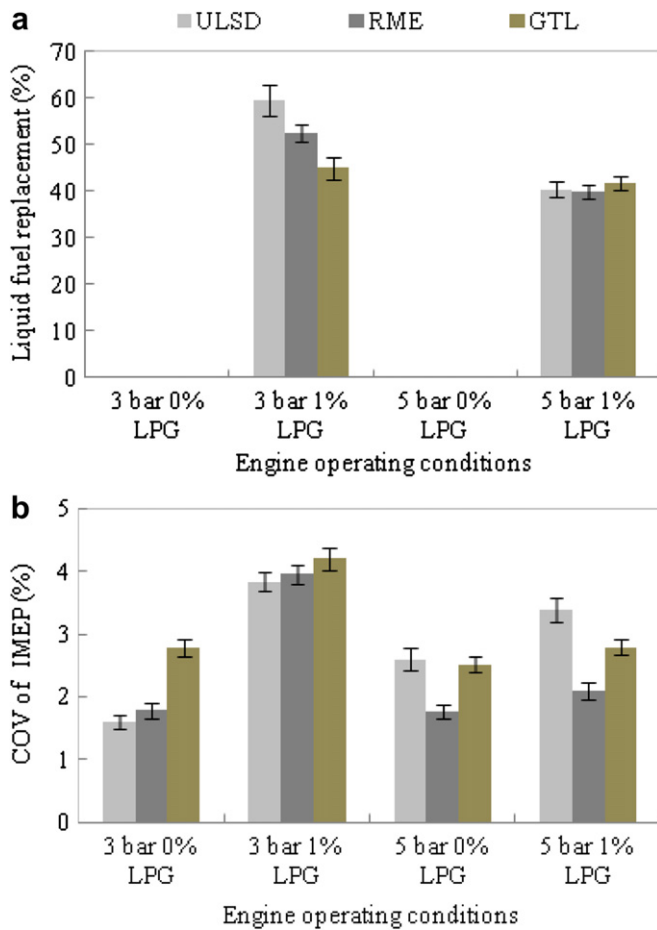


Fig. 10. Liquid fuel replacement and COV of IMEP for different liquid fuels.

cylinder charge. When the effect of fuel is considered it was obtained that LPG addition to GTL at low engine load caused an increase in COV of IMEP but in all the cases COV was maintained in the acceptable range. According to Heywood, the driveability problems are likely to occur when the value COV of IMEP is more than 10% [21].

There is no significant difference between the engine thermal efficiency of RME and ULSD, as it was widely reported in the literature [30–33] and the trend in the baseline conditions (0% LPG

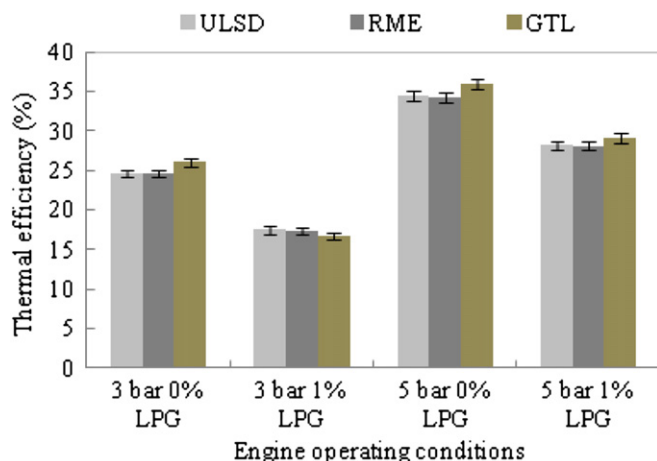


Fig. 11. Brake thermal efficiency for different liquid fuels.

addition) was kept similar to the dual fuel mode. GTL combustion showed better engine thermal efficiency compared to that of ULSD and RME. This gain is due to the improved fuel atomisation and the high CN of GTL which led to shorter ignition delay [34]. In addition, enhanced complete fuel oxidation reduces fuel consumption and improves thermal efficiency [10]. However, in dual fuel mode in low engine load and 1% LPG addition, the engine thermal efficiency in LPG–GTL dual fuelling was reduced because of the retarded combustion as a result of the high compressibility and low density of GTL. This caused lower in-cylinder temperatures and reduced the indicated work.

### 3.2.2. Emissions

At engine operating conditions without LPG, the combustion of RME and GTL produced lower HC and CO emissions compared to those of ULSD fuelling (Fig. 12). There are several factors that affect both formation and oxidation of these emissions. One of them is the oxygen content in RME which improves HC and CO oxidation [35]. Both higher H/C ratio and CN of GTL (Table 2) compared to ULSD can help hydrocarbon fuel oxidation. However, with 1% LPG addition, HC emission was increased in the GTL dual fuelling. This result indicates a poor oxidation of LPG in GTL dual fuelling mainly due to less energy released and low combustion temperature.

Soot concentration was gradually reduced as the fraction of LPG in the cylinder increased for all engine operating conditions and liquid fuels studied. As it is explained previously, the main reason for this soot reduction is the decrease of the number of liquid fuel rich regions in the combustion chamber.

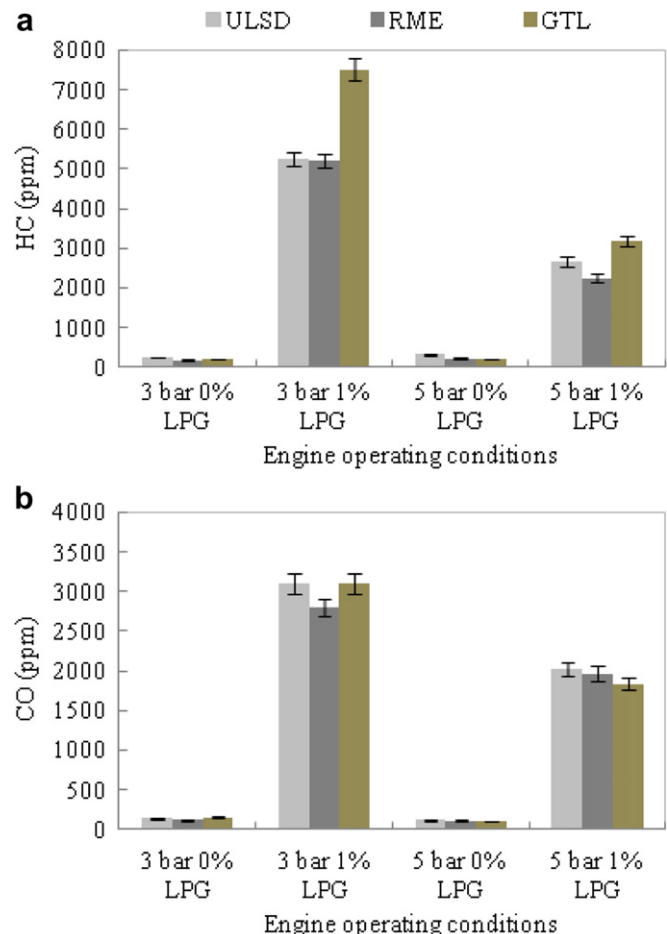


Fig. 12. Hydrocarbon and carbon monoxide emissions for different liquid fuels.



At low engine load and 0% EGR,  $\text{NO}_x$  emissions also decreased when LPG was inducted, breaking the soot- $\text{NO}_x$  trade-off for all the liquid fuels combustion (Fig. 13a). In 20% EGR, a simultaneous soot and  $\text{NO}_x$  reduction with LPG application was also obtained (Fig. 13b). GTL showed better soot- $\text{NO}_x$  trade-off lines with the application of EGR than in the case of ULSD (i.e. lower  $\text{NO}_x$  and comparable soot emissions) at each percentage of LPG addition. The lower  $\text{NO}_x$  emissions in the case of GTL (paraffinic fuel) are the result of a slower rate of premixed combustion (see Fig. 9) and lower in-cylinder temperature, a consequence of its high CN. On the other hand, the large proportion of fuel which is burned in the diffusion combustion phase enhances soot formation. However, its free-aromatic composition balances soot formation and hence produces similar soot emission to ULSD. Therefore, LPG-GTL dual fuelled engine mode is a good combination as the high CN of GTL can partially compensate the poor auto-ignition properties of LPG.

In the case of RME, the oxygen-assisted and slightly advanced combustion caused a faster combustion rate in the premixed combustion phase (Fig. 9). These increased the in-cylinder pressure and temperature and consequently engine-out  $\text{NO}_x$  emissions compared with ULSD and GTL. Soot emission for RME was the lowest mainly due to the oxygen content which reduces soot formation and enhances soot oxidation [13,14,30,32,36]. The application of EGR in 1% LPG produced a significant reduction in  $\text{NO}_x$  without increasing soot emission. Based exclusively on soot and  $\text{NO}_x$  emissions (without taking into account the engine thermal efficiency, HC and CO penalties) 0% and 20% EGR for GTL and 20% EGR for RME with 1% LPG are the optimal combinations.

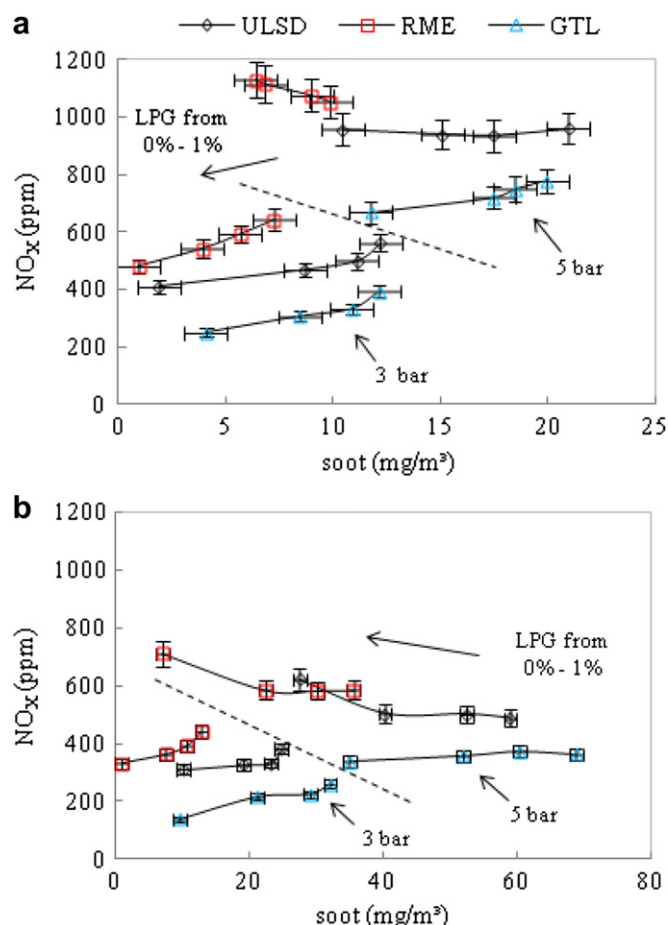


Fig. 13. Soot- $\text{NO}_x$  trade-off for different liquid fuels. a) 0% EGR, b) 20% EGR.

At high engine load and 0% EGR, a simultaneous soot and  $\text{NO}_x$  reduction was only seen with GTL dual fuelling. In the case of RME,  $\text{NO}_x$  gradually increased with LPG addition, limiting the use of LPG-diesel dual fuel combustion at high engine load without EGR. However, at 20% EGR a simultaneous reduction in soot and  $\text{NO}_x$  were obtained for GTL and RME. Comparing fuels, GTL evidenced a better EGR tolerance at each LPG concentration than RME. In the case of 1% LPG and RME, the implementation of EGR reduced  $\text{NO}_x$  emissions without any soot penalty. Therefore for RME, it is suggested a combination of EGR and LPG addition to optimise LPG-diesel dual fuel combustion.

LPG addition for RME and GTL produced lower SOM (Fig. 14a) than in the case of liquid only fuel combustion, despite the significant increment in HC emission (Fig. 14b). As in the case of ULSD, this is due to the decrease in soot emission and the lower dew point of unburned hydrocarbons when LPG is added.

RME produced higher SOM emissions than in the case of ULSD with and without LPG addition (Fig. 14a). This trend can be justified by the lower volatility of the hydrocarbons emitted using RME [28]. Additionally, the high viscosity and density of RME led to the bigger fuel size droplets and slower evaporation rate, which could tend to raise SOM emission [35]. The low soot emission (Fig. 13) but high SOM for RME resulted in higher SOF compared with ULSD and GTL (Fig. 15). Generally, SOM emission with GTL is higher than that of ULSD specifically at high engine load. This is due to the higher T90 (distillation temperature at 90 °C) of GTL and a lower in-cylinder pressure and temperature compared with ULSD (see Fig. 9) [37].

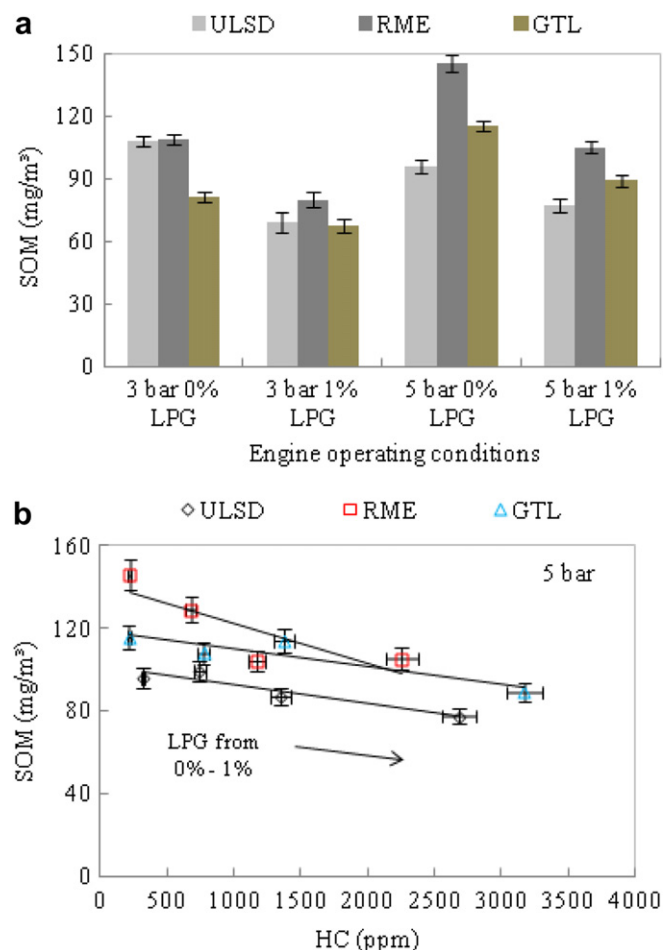


Fig. 14. Soluble organic material (SOM) and HC-SOM correlation for different liquid fuels.

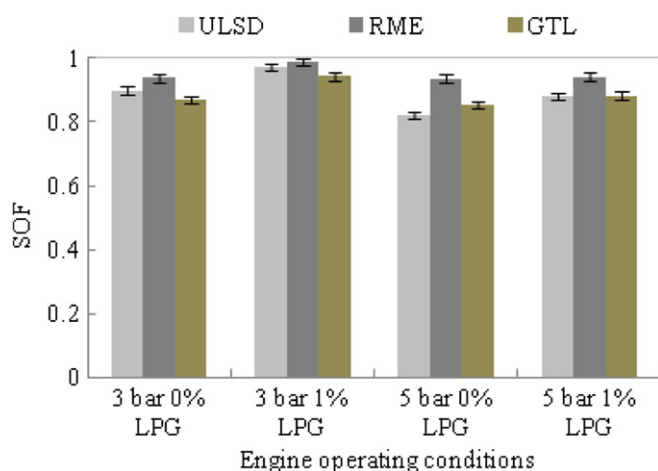


Fig. 15. Soluble organic fraction (SOF) for different liquid fuels.

On the other hand, the absence of aromatic compounds in GTL reduces soot formation, resulting in a higher SOF compared with ULSD. As in the case of ULSD, SOF increased for RME and GTL at low engine load compared with high engine load, as it was previously explained.

#### 4. Conclusion

The combustion of inducted LPG to diesel combustion significantly increases the gaseous HC and CO emissions as a result of its low CN, replacement of the oxygen concentration and possibly due to problems associated with mixing air and gaseous fuel. Increased LPG concentration led to a simultaneous reduction in soot and  $\text{NO}_x$  emissions.

Different types of diesel fuels used have shown to have no influence on gaseous HC and CO emissions trends but the substitution of the diesel fuel with RME or GTL resulted in significantly reduced  $\text{NO}_x$  – Soot emissions trade-off line to lower values, with the reduction following the order of  $\text{GTL} > \text{RME} > \text{ULSD}$ .

The results of the LPG–RME dual fuelling have shown that EGR would be beneficial at high engine load conditions while in the LPG–GTL dual fuelling optimisation of the GTL fuel injection characteristics can enhance further low engine load to keep the engine emission benefits.

Overall, the study confirms that diesel fuel properties need to be optimised for the different LPG additions in order to obtain the improved engine-out emissions of LPG–diesel dual fuelling.

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#### Nomenclature

BS	British standard
CAD	crank angle degree

CN	cetane number
CO	carbon monoxide
$\text{CO}_2$	carbon dioxide
COV	coefficient of variation
DC	diffusion-charging
DI	direct injection
EGR	exhaust gas recirculation
FID	flame ionization detector
GTL	gas-to-liquid
$\text{H}_2\text{O}$	water
HC	hydrocarbon
IMEP	indicated mean effective pressure
LCV	lower calorific value
LPG	liquefied petroleum gas
$\text{N}_2$	nitrogen
$\text{NO}_x$	nitrogen oxides
PM	particulate matter
RME	rapeseed methyl ester
ROHR	rate of heat release
rpm	revolutions per minute
SOC	start of combustion
SOF	soluble organic fraction
SOI	start of injection
SOM	soluble organic material
ULSD	ultra low sulphur diesel

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